Metal fatigue due to excess vibration and dynamic stresses on a hydro power plant

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The normal life of most of equipment cannot be more precisely defined than some tens of years. It depends upon normal and abnormal stresses acting on it. If by proper routine and capital maintenance, these stresses are kept with in limits, then life can be increased to 50-60 years and even more. Therefore regular monitoring of vibration from start to permissible over load conditions is very urgent. Precisely the same has been done successfully in all the 28 machines in 6 power plants with an installed capacity of 2771 MW of BBMB. These machines were commissioned from 1954 onwards. Whenever any fault developed in any machine, instead of simply attending it, diagnostic tests were carried out and stresses acting on machine were analyzed. This is the main reason for their trouble free service. These studies were carried out at that time when this concept was not well known. The author had the privilege of working and participating in such tests for about 14 years. This article is based upon all such tests.

1. GENERAL INTRODUCTION

1.1 The dynamic behavior of a machine is a very complex phenomena & its complete explanation is beyond the scope of this article. Complete book can be written on this subject. However sufficient introduction for understanding the behavior has been dealt here. From dynamic stresses point of view, the operating regime of the machine can be divided into following -

i). Steady state operation
ii). Transient state operation.
iii). Resonance of components

1.1.1 STEADY STATE OPERATION

During steady state operation, following forces act on a machine

i). Static weight including water weight. This does not need further explanation.

ii). Residual stresses. These may be initial residual stresses created at the time of construction & assembly. During operation, these may be generated during weld repairs & non-uniform heating due to which stress locking is caused. The effect of residual stress is pronounced on intricate shapes or part of a component.

iii). The most important is dynamic forces generated due to rotation of unit. During steady state, unit operates at a constant head, speed, load, wicket gate opening etc. The forces acting during this period are constant or periodic in magnitude, direction & frequency. However due to defects like excess pressure pulsation or cavitation or misalignment, non periodic forces of random nature having different direction, amplitude & frequency also appear.

1.1.2 TRANSIENT STATE OPERATION

This is state, when head or load or wicket gate opening changes i.e. starting, synchronizing, increase in load, decrease in load, stopping, tripping, faults on machine, braking of machine, run away speed etc. During this period, vibrations do not follow any set pattern. These change in magnitude, direction or frequency. Their values depend upon increase or decrease in quantity of inlet water. Amongst the humane errors in introducing unnecessarily unsteady state stresses is rapid increase / decrease in load or frequent starting / stopping; bad & worn out governor causing severe hunting & giving rise to varying perturbation forces on blades & wicket gate etc.

1.1.3 RESONANCE OF COMPONENTS

Runner, penstock, spider, generator stator, head cover and shaft system are effected by resonance.

2. FATIGUE FAILURE

2.1 Thus during operation, various steady & unsteady forces act on a hydro unit & its various components. Unsteady state forces super impose on steady state forces. These forces give rise to vibrations - singularly or collectively. Since various components are fixed to each other, vibration due to one component is transferred to other components & so rigidity of each component has to be ensured. These vibrations result in deformation of different components, which in turn cause additional stresses on effected parts. These stresses are proportional to deformation or deflection. The unit or its components are designed with sufficient safety margins to withstand normal stresses. But high vibrations & dynamic stresses produced thereof, result in gradual development of hair cracks. If no remedial measures are taken in bringing down these vibrations & attending these surface or subsurface hair cracks, then failure of effected components occurs abruptly without any further notice. This process of failure is called Fatigue Failure.

2.2 Excess vibrations & resultant stresses in the effected parts cause excessive fatigue & hence failure of that part. Vibrations from any one source may induce more vibrations on other sources & the net effect may be more severe & cascading in nature. Excess vibrations may also effect foundations of machines or part of building, if their resonant frequency coincides with one of the exciting machine frequencies & causes excess vibrations. If the value of vibration & pulsation are within limits, than these do not affect machine life by themselves.

The design of machines manufactured during fifties or sixties was liberal & fatigue failure due to vibration stresses was rare. However, now due to increase in size of machines & reduction of safety factors, dynamic rigidity of the structures has been reduced which has increased risk of fatigue failure. During design, amplitude of expected vibrations are taken into account & with the present day precision instruments; it should be possible to properly monitor these vibrations. Therefore regular monitoring of vibration from start to permissible over load conditions is very urgent.

There may be many sources of vibration but real problems may be the creation of only one or two sources. If it does not become possible to restrict these during whole operating regime, than the safe operation regime can be defined. If vibration level is beyond permissible limits, then first of all, sources contributing to these vibrations have to be identified & efforts made to eliminate these. It may not be possible to completely eliminate these vibrations but efforts have to be made to bring them within safe limits.

2.3 The strength & thickness etc. of the material should be so chosen that it is able to with stand repeated action of variable forces. This strength is called fatigue strength or endurance of material. In a well designed (with sufficient factor of safety) & constructed machine, normal load, vibration & hence stresses do not adversely effect mechanical parts. If the operating stresses are below designed stresses, these components have a long life. However, due to excess vibration & resultant stresses, fatigue failures occur. Case of failure of trailing edges of Francis runner is cited. The calculated absolute pressure & stresses on the trailing edge is quite small (Fig. 11.2). Therefore, for the sake of economy, mechanical rigidity, cavitation coefficient & efficiency, the thickness of out let edge is kept small. But many a times, due to pressure pulsation, there is a reversal of stresses at the outer edge, which results in appearance of cracks, which are removed by increasing thickness & or welding additional plate.

2.4 The hydro turbines unlike other machines like turbo sets, motors etc. are tailor made to suit specific requirements of water head, quantity, space limitation etc. Its size & weight is huge & in-accuracies during design, construction & assembly can take place. Since these also operate under varying head & quantity of water, there operational behaviour like vibrations, pressure pulsation etc thus cannot be very accurately predicted except some through model testing of turbine. Even after model testing, their value on prototype can be reduced but not eliminated.

2.5 Excess vibrations causes excess wear & tear & fatigue failure of
a). Runner blades, guide vanes, rim, bearing, shaft, runner labyrinth, shaft seal etc.
b). Shearing or loosening of different bolts & nuts at all effected locations. Many a times, these lead to excess bearing gaps & rapid wearing of guide bearings - needing frequent replacement of bearing pads or damage to bearing journal.
c). Loosening of stator core bolts, wedges, stampings, pole wedges etc.

2.6 Excess vibration due to inadequate fastening of core, various metal covers & fixers, turbine covers, pressure pulsation & cavitation etc. give rise to excess noise.

3 VIBRATIONS

Vibration is a phenomena of rotating machines. It is a periodic motion around an equilibrium position. With increase in defects, the vibration level increases. Each defect has its own characteristics. Therefore by analyzing the vibrations, the type of defect can be determined.

3.1 GENERAL GUIDELINE FOR MEASUREMENT OF VIBRATIONS

i) At present, very few standards of vibrations are available. The vibrations rise with increase in size of machine & its maximum value should decrease with increase in speed. Its measurements & correct interpretation requires great experience. Following types of vibrations are measured:-

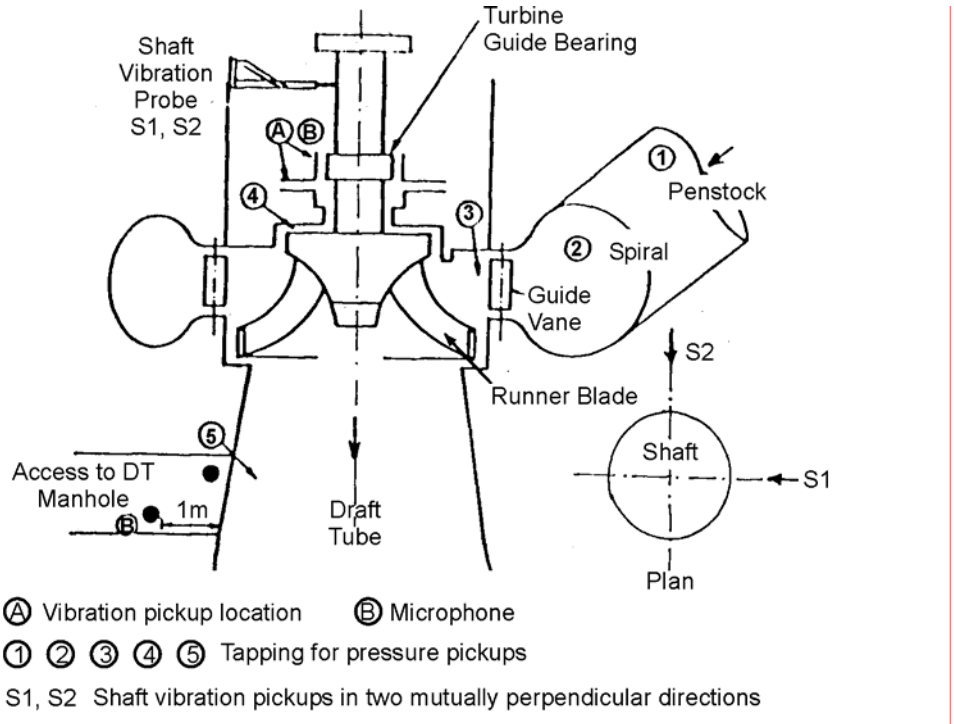


FIG. 11.1. LOCATION OF PICKUPS FOR HYDRO POWER PLANT MEASUREMENTS

-Displacement vibrations
-Velocity vibrations
-Acceleration vibrations
-Phase vibrations
-Frequency vibrations

ii) Since the speed of hydro generator is less, the vibrations should be measured with a low cycle oscillator. It should be contactless relative pickup type. The pickups work on the principle that when there is a change in the gap between the magnet and the shaft, it causes a change in the E.M.F. of the winding of pickup. The sensitivity of pickup on an average is equal to 0·5 MV/micron.

iii) Machine should operate under design parameter.

iv) Measuring instruments should be accurate.

v) Guide Vane opening should be checked from servo motor stroke & should be accurate to $\pm$ 1%. Similarly blade angle measurements of Kaplan turbine should also be accurate to $\pm$1%.

vi) **Shaft vibrations -** Shaft vibrations are measured at different locations along its axis. Conventionally only shaft run out was measured with the help of dial gauges. This does not give clear picture of shaft centerline motion or static shift of shaft position under various operating conditions. Shaft vibrations are measured by mounting non contact probes at all the bearings in two mutually perpendicular directions, usually one in the

down stream direction and other in the direction perpendicular to it. The overall shaft vibrations are noted and their signal is spectrum analyzed to study the dominant frequencies. The signal in two directions is plotted in X and Y-axis to obtain a picture of shaft orbit at that particular bearing through which it is possible to assess the condition of bearing. Maxm. value is 200 microns (pk to pk)

vii) **Bracket Vibration -** Seismic probes (Contact type probes) are mounted on the bearing bracket viz TGB, LGB and UGB which give acceleration, velocity and displacement signal at particular bracket. The spectrum analysis of such signal indicates high frequency components, which cannot be detected by shaft vibration monitoring. Maximum value for medium speed turbine is 150 microns (pk to pk)

iiv) **Noise Analyses -** Noise at hydro stations is due to flow of water, cavitation and vibration. For measuring noise level, monitors are installed at various locations. For cavitation, these are installed near draft tube manhole. Maximum value any where is 90 dba.

ix) At the time of measuring vibrations also measure clearance of various guide bearings, noise level & pressure pulsation.

3.2 SOME STANDARDS IN MICRONS (P.K-P.K) are as follows –

Speed RPM	MAXIMUM FRAME VIBRATIONS				**MAXIMUM SHAFT DISPLACEMENT**		
	J. H. Walker's Book*		As per VDI2059 Part-I	B.S.2613 Value **	T.P.E. Practice		VDI-2059**
	Smooth Value	Fair Value			General value	Max.	
166.6	70	170	N.A.	-	150	200	170
200	62	160	200	-	N.A.	N.A.	155
300	50	150	N.A.	125	N.A.	N.A.	125

** Published by BHEL. ** Derived from graph*

ii) **Hitachi's recommendations** for 166.6 & 300 rpm turbines at Bhakra left, Ganguwal & Kotla power houses-

LOCATION	**NORAML VALUE**		**MAXIMUM VALUE**	
	166.6 rpm	**300rpm**	**166.6rpm**	**300rpm**
Bearing Support	< 270	< 225	< 450	< 375
Draft Tube	< 30	< 30	< 50	<50
Shaft vibration/bearing gap	< 40%	< 33%	< 56%	<50%
For example, vibrations with 0.2 mm gap.	<80	<60	<112	<100

iii). **B.HEL'S** recommendations for their different machines (maxm. pk-pk)

Location	**Ranjit Sagar P.P 4X150 MW, 166.6 RPM**	**(Maxm.Peak to Peak) Dehar P.P ;;6 x 165 MW 300 RPM**
Bearing bracket	120	100
Slip ring	NA	0.2 mm
Shaft displacement	60 to 80% of diametrical G.B. clearances	

iv**) Vibration in Microns (pk-pk) as per Russian practices** *(Book on vibration of hydro units in hydro electric power plants by L.A.Vladislavlev)*

S.No.	Speed	Excellent	Good	Satisfactory	Poor
1	62.5	0-50	50-100	100-160	.160
2	150	0-40	40-90	90-140	.140
3	187	0-40	40-90	90-140	.140
4	214	0-30	30-80	80-130	.130
5	250	0-30	30-80	80-130	.130
6	300	0-20	20-70	70-120	.120

v) Some General Practices for Vibrations

a). Vertical vibration ----------0.2 mm

b). Run out at slip ring & commutator----0.2 mm

c). Maximum shaft displacement PK-PK-----60-80% of diametrical gap clearance at a location on the shaft as close to guide bearing as possible.

vi) **Velocity & Acceleration**

Measurement of velocity & acceleration of vibration to assess the dynamic behavior is also very necessary. The additional force generated due to vibration-

P = M x A where M is Mass & A is acceleration. General values are as follows:-

a) Vrms in mm /second for large machines up to 300 rpm.----8mm/sec.

b) Vrms in mm/ second for 10 to 200 rpm machines as per VDI 2056

< 1.8 mm /second	Good
1.8 to 4.5 mm/ second	Satisfactory

vii). MAXIMUM NOISE LEVEL SHOULD BE AS UNDER --

Generator stator	< 85 dbA 1 Mtr from frame
Turbine at T.G.B., Spiral casing, Draft tube	< 100 dbA

iix) Measure deflection of load bearing bracket of stationery & running machine with the help of dial gauges placed 90^0 apart & compare it with commissioning / manufacturer's data. It would give an idea of any increase in axial hydraulic thrust.

ix). The detailed analysis of vibrations & remedial measures requires detailed knowledge of machine behavior. It would be worthwhile to measure it at least annually & discuss the measurements with manufacturers experts once in 2-3 years. Similarly once in 2-3 years, readings on one machine may be got taken under the manufacturers experts supervision or independently by them along with their comments. The expenditure on this account would be very insignificant as compared to advantages. This would also enable field engineers to update their knowledge & see the latest equipment. Besides, for detailed analysis purposes, 5-6 vibration meters are required for installing at different locations & being costly, this quantity is not available in any plant.

ix**) On line vibration monitor**:- For silt effected power plants , where vibrations increase rapidly due to various factors, on line vibration meters can be installed. These have following advantages:-

-Keeping machine in operation till vibrations level is with in limits.

-The degradation is limited as after knowing increased trend, the machine can be stopped/ planned for shutdown.

3.3 Sources of vibrations

The vibrations can be due to following main sources:-

i) Hydraulic Vibrations
ii) Mechanical Vibrations
iii) Electrical Vibrations

These are measured in displacement (microns Pk-Pk), velocity (mm/sec.) & acceleration (m/sec^2) modes at different points viz draft tube (non concreted portion), top cover, T.G.B., LGB, UGB & shaft. Vibration at all these locations be measured from no load without excitation up to permissible over load. These values give an idea of the forces (Hydraulic, Mechanical & Electrical), which are being transmitted to bearings & the remedial measures required.

4 HYDRAULIC VIBRATIONS (Fig.11.3)

Due to flow of water through turbine, runner, scroll case, guide & stay vanes etc., eddies are produced. The disturbance caused by these eddies not only cause vibrations & stresses on runner blades, but these are also transmitted to water passage wall & equipment viz. scroll case, top covers, wicket gates, draft tubes etc. & cause these to vibrate. Metal fatigue of runner is mostly due to such excess hydraulic stresses. The effect of mechanically generated induced vibrations (mechanical & electrical) causes comparatively very less effect on runner blades. Similarly the effect of hydraulically induced vibrations is very less on generator. This is due to damping effect of large mass of rotor. There are many causes of hydraulic vibrations but major are as follows

i). High dynamic stresses due to improper turbine design.

ii). Non uniform flow & vortex formation.

iii) Matching natural frequency with von-Kerman vortices formation.

iv). Excess wears & tears.

v). Cavitation phenomenon *(refer para 2.2 chapter 2)*

vi). Runner imbalance

4.1 HIGH DYNAMIC STRESSES DUE TO IMPROPER TURBINE DESIGN

This is explained with the help of following case studies

4.1.1 CRACKS IN PONG POWER PLANT MACHINES –

6 X 60 MW. (Now 6 x 66 MW)i). Huge cracks developed on first 3 units of Pong power house within 2 years of commissioning as described below

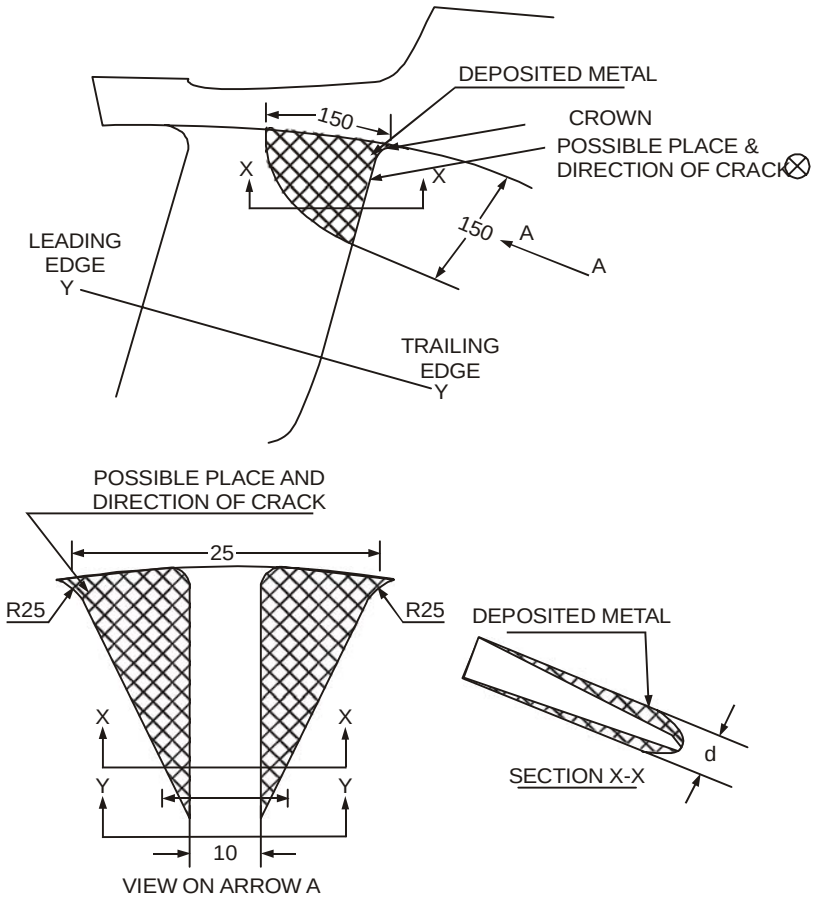
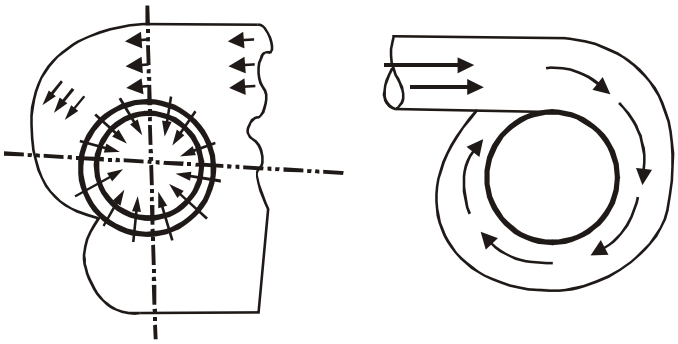
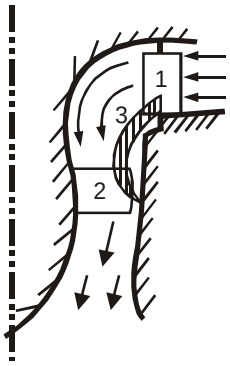


FIG. 11.2-MODIFICATION OF RUNNING BLADE OUTLET EDGE NEAR CROWN OF PONG POWER PLANT

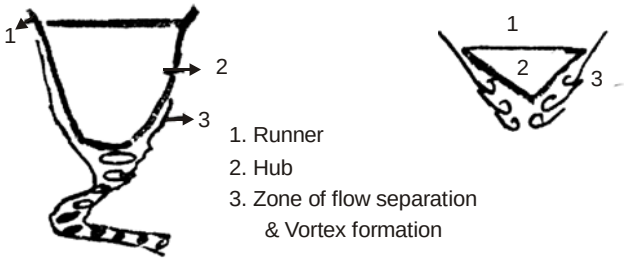
Unit No.	Date of commissioning	First Inspection			Second Inspection			Third Inspection		
		Date	Period Run Hrs	Result	Date	Period Run hrs	Result	Date	Period Run hrs	Result
1	20.01.78	4/78	1983	OK	3/79	9148	OK	12/79	14972	Major Cracks found
2	30.03.78	9/78	3340	OK	3/79	7770	OK	12/79	13221	-do-
3	26.10.78	FIRST INSPECTION →						12/79	7800	-do-



Non uniform velocity distribution in spiral casing of turbine

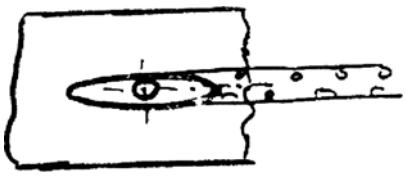


Separation of vortex filament
1. Guide Vane 2. Runner Blade
3. Vortex filament

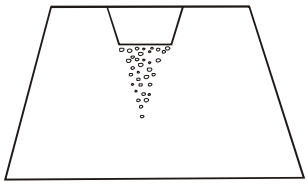


1. Runner
2. Hub
3. Zone of flow separation & Vortex formation

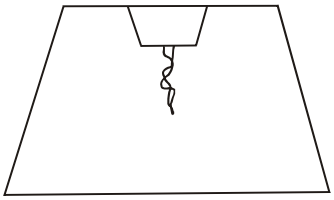
INFLUENCE OF SHAPE OF BODY IN FLOW ON VIBRATION OF TURBINE



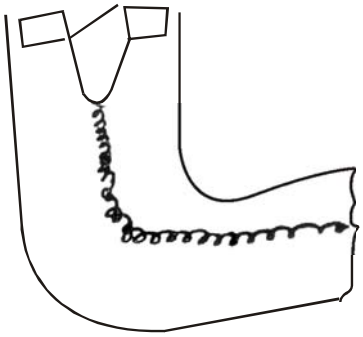
VORTEX TRAIL SHED FROM BUTTERFLY VALVE



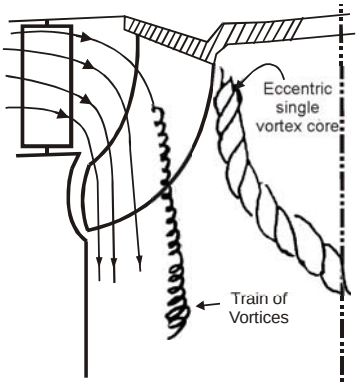
A turbine operation w/o any vortex or flow separation but a few air bubbles near the bottom of runner cone



A turbine w/o vortex and flow separation but a small size of rope formation



A turbine with vortex tail shed in a draft tube



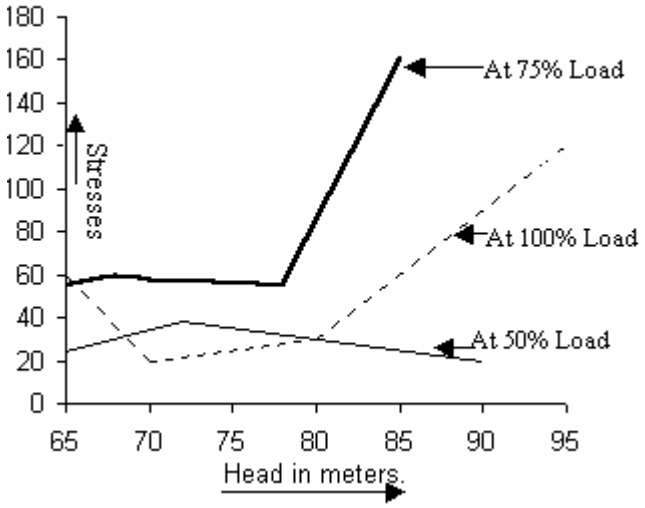
Typical flow in runner and draft tube at part gate opening

FIG. 11.3 A ROUGH ILLUSTRATION OF HYDRAULIC VIBRATIONS

In the case of first two units, major cracks were observed on almost all blades. These were found varying from 90 to 995 mm. Cavitation had also occurred. Incase of unit no. 3, major cracks were found on two blades.

ii). Investigation was conducted with the help of BHEL. Inspection revealed neither pitting nor erosion. Chemical composition & mechanical properties appeared in order (as tabulated under chapter 13 para 10). Casting was also found satisfactory. Further study revealed that cracks were owing to fatigue failure, which was not due to normal operating stresses but due to higher vibration stresses. Model test revealed that these dynamic stresses increased very sharply at a head higher than 85 mtr (fig.11.1). The available head due to water level in reservoir varies from 47.85 mtr. to 95.1 mtrs & the machine is designed for a net head of 68.58 mtr. The cracks in all the three units were observed after high head period. Therefore the results of model tests that stresses increase sharply after a head of 85 mtr. were accepted. (Fig.11.2).

Fig.11.1 Dynamic stresses on runner blade material



iii) Based on above finding, BHEL recommended following

a) To carry out frequent inspection of runner as follows to detect & rectify any incipient damage in early stages -

Heads above 80 Mtrs	Every 500 hrs. of operation.
Heads between 65-80 Mtrs	Every 2000 hrs. of operation.
Heads below 65 Mtrs	Every 3000 hrs. of operation.

The inspection was suggested to be done by dye penetrate & ultrasonic tests for detection of micro (Surface & subsurface) cracks which otherwise are not visible to naked eyes.

b). Since the dynamic stresses beyond 85 mtrs. increases sharply, BHEL recommended that blade thickness at outer edges near the crown should be increased to reduce their effect. Similarly to avoid stress concentration at junction of crown & outlet edge, the fillet radii at the junction be also increased (Fig. No.11.2).

B.B.M.B. Engineers are implementing para a regularly & after carrying out modifications as described in this paragraph, no problem has been encountered since 12/79. This has also reduced vibrations & noise level.

4.1.2 BHAKRA RIGHT BANK POWER PLANT - ORIGINAL 5 X 120 MW (Now 5 x 157 MW)

i) Five units were commissioned between 5/1966 & 12/1968. The problems of cracks, cavitation & washing awayof blades was experienced from the beginning. Investigations revealed fatigue failure due to following design defects

a) Long thin outlet edge of runner blade (8 mm only)

b) Pressure pulsation near runner outlet & surges in draft tube.

c) Shorter draft tube depth.

d) Possibility of closeness of some natural frequencies with von-Kerman vortices formation.

ii). Following remedial measures were taken under suppliers & BHEL Engineers guidance**(**Fig.11.4)

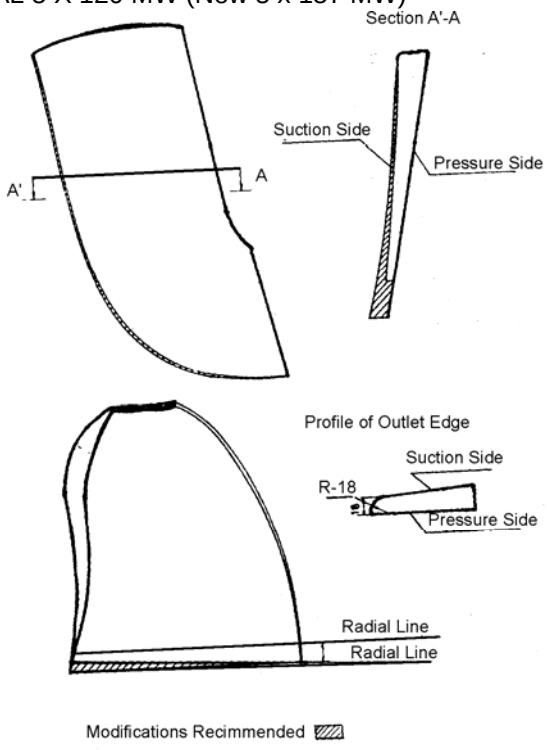


FIG. 11.4 MODIFICATION OF RUNER BLADES OF BHAKRA RIGHT BANK P.P.

UNIT NO. 6 & 7 (THESE ARE OF CAST STEEL)

1969-70-71 The trailing edges of the runner blades (about 200 to 250 mm width) were

replaced with stainless steel plates supplied by manufacturer.
1971-72-73: Rounding of the trailing edges of all the runner blades on pressure side.
1973-74: Machine no. 6 only - Thickness of leading edges of all blades increased by depositing metal.
1980-81: Trailing edges of all the blades were trimmed by 43-91 mm with subsequent rounding on suction side.

UNIT NO. 8,9 & 10 (MADE OF STAINLESS STEEL)
1970-71-72: Strengthening of all runner blades was carried out by welding 5-6 mm thick stainless steel plate in width of 150-200 mm from upper to lower end of trailing edge. The designer had in fact suggested a plate of 6-12 mm thickness, which was not carried out due to fear of distortion of blades.
1973-74: Rounding of the trailing edges of runner blades 50 mm radius on suction side was carried out.
iii) ALL THE MODIFICATIONS WERE CARRIED OUT. NO PROBLEM HAS BEEN EXPERIENCED AFTER THAT EXCEPT ON UNIT NO. 7, WHERE THICKNESS OF LEADING EDGES OF BLADES COULD NOT BE INCREASED. IT IS WORTH MENTIONING THAT THERE ARE MANY SIMILAR CASES OF REDUCTION IN DYNAMIC STRESSES AND VIBRATIONS DUE TO THE IMPROVEMENT OF TRAILING EDGES OF BLADES IN INDIA AND ABROAD.

4.2 NON-UNIFORM FLOW AND VORTEX FORMATION (Fig.11.3)

4.2.1 The scroll case, guide vanes, runner & draft tube are so designed & assembled that no eddies are produced when water either enters or leave turbine. Excess eddies if produced, cause excessive pressure, pulsation, shaft torque pulsation, rotation speed pulsation, power pulsation etc. etc. These increase vibrations & noise, which causes fatigue failure of all turbine components in general but runner blades in particular. It is not necessary to measure all these pulsations. The most dominant & important is pressure pulsation, which is the ratio of oscillatory variation of pressure at a given time interval to the net head. This oscillatory value is the algebraic difference between peak-to-peak values. In case of sinusoidal quantity, it is twice the amplitude. The Pressure pulsation should remain with in $\pm$6% at important locations. Effect of flow of water is described as below—

i) FLOW OF WATER IN RUNNER BLADES
Bad design & constructions results in variation of velocity in different parts of scroll case, at connection with stay ring & at entrance to guide vanes (Fig.11.3) & runner. This results in asymmetrical flow before the runner, giving rise to varying forces exerted on each of its blades, vibrations & hence stresses. Similarly due to creation of vortices from the trailing edges & runner blades, the excessive pressure pulsation occurs. Such cases are tackled by changing the trailing edges of blades (case study 4.1.2).

ii) FLOW OF WATER THROUGH VANES AND WICKET GATES
Stay vanes & wicket gates interfere in the uniform flow of water.Alone or coupled with non-uniform flow due to scroll case as described above, it causes separation of layers of water from the trailing to leading edges of stay vanes & wicket gates. This results in the formation of vortices (fig.11.3). These vortices may reach the runner blades & cause vibrations & pressure changes on the runner blades, resulting in excess stresses on the blades. If the natural frequency of this flow comes close to runner blades, then it causes fatigue failure of the blades. If it comes closure to any other component of the unit, it causes high amplitude vibrations in that component. Since it is difficult to modify / redesign stay vanes or wicket gates, the remedial measures are taken by improving mechanical rigidity of runner blade or any other effected components.

iii) FLOW THROUGH DRAFT TUBE
At low loads, due to reduced wicket gates opening, there is comparatively less pressure at runner inlet but due to full speed, the centrifugal force is as per rated output. This results in formation of vortices below runner blades. The size of draft tube corresponds to rated load but quantity & pressure of water being less, a dead water zone is created in draft tubes, giving rise to another vortices (fig11.3) As the load increases, these vortices join & take the form of a rotating spiral & flow through draft tube. This results in non-uniform flow, noise, pressure pulsation & torque fluctuation in draft tube which travels towards runner & even scroll case & cause excessive vibrations & metal fatigue of effected components. These are more in short draft tubes (case study 4.1.2). It is reduced by injecting compressed air, providing cross pieces & splitter vanes in the draft tube for breaking the vortex. The air is delivered in the central portion of draft tube below the runner. For forced air, pipes either through shaft or draft tube lining are laid. For delivering atmospheric air, valves erected on turbine head cover are installed. These open & close automatically when vacuum is created.

iv) PRESSURE VARIATION

The maximum momentary pressure on the high pressure side & minimum momentary pressure on the low pressure side of a turbine normally occur during the shut down operation when a specified load is reduced to zero. The minimum momentary pressure on the high pressure side & maximum momentary pressure on the low pressure side occur during opening movement of the shut off device from 0 to full load opening.
Maximum variation = $\pm$ 100 $p^{-0.25}$ where p is gauge pressure in Pascal.

4.2.2) By carrying out field tests of pressure pulsation, vibration & noise; the reason for excessive vibration can be analyzed & remedial measure taken. Worst regime of operation can also be avoided. It is generally monitored at four points –

i) Before main inlet valve.

ii) After main inlet valve.

iii) Between guide vanes & runner.

iv) In draft tube cone.

However many a times, provision does not exists for measurement at all the points. In that case, minimum requirement i.e. pressure at draft tube be monitored. Pressure pulsation in draft tube should not exceed +-5% of the head. Pressure pulsations of a typical Francis turbine at draft tube at different load are illustrated in fig.11.5.The phenomenon is explained with the help of following case studies.

4.2.3 CASE STUDIES

4.2.3.1 PONG POWER PLANT 6 X 60 MW

i) As described under para 4.1.1. there was excessive blade cracking on unit no. 1, 2, 3 at Pong power plant. The initial noise level near draft tube manhole on all machines was 100-112 dba, vibrations at draft tube cone were up to 92 microns, oscillation of power & current were +-2 MW & +-100 Amps respectively & cracking of draft tube cone liner near manhole was occurring. All these were high. BHEL carried out draft tube pressure pulsation measurement on the prototype at site on all machines at different heads & loads. It opined that up to a head of 80 Mtr; pressure oscillations are not causing any damage. The noise level & oscillation increase beyond 80 Mtr. The study could not be completed, as higher head were not available at that time. Based on the above findings, following methods for improving runner aeration were tried

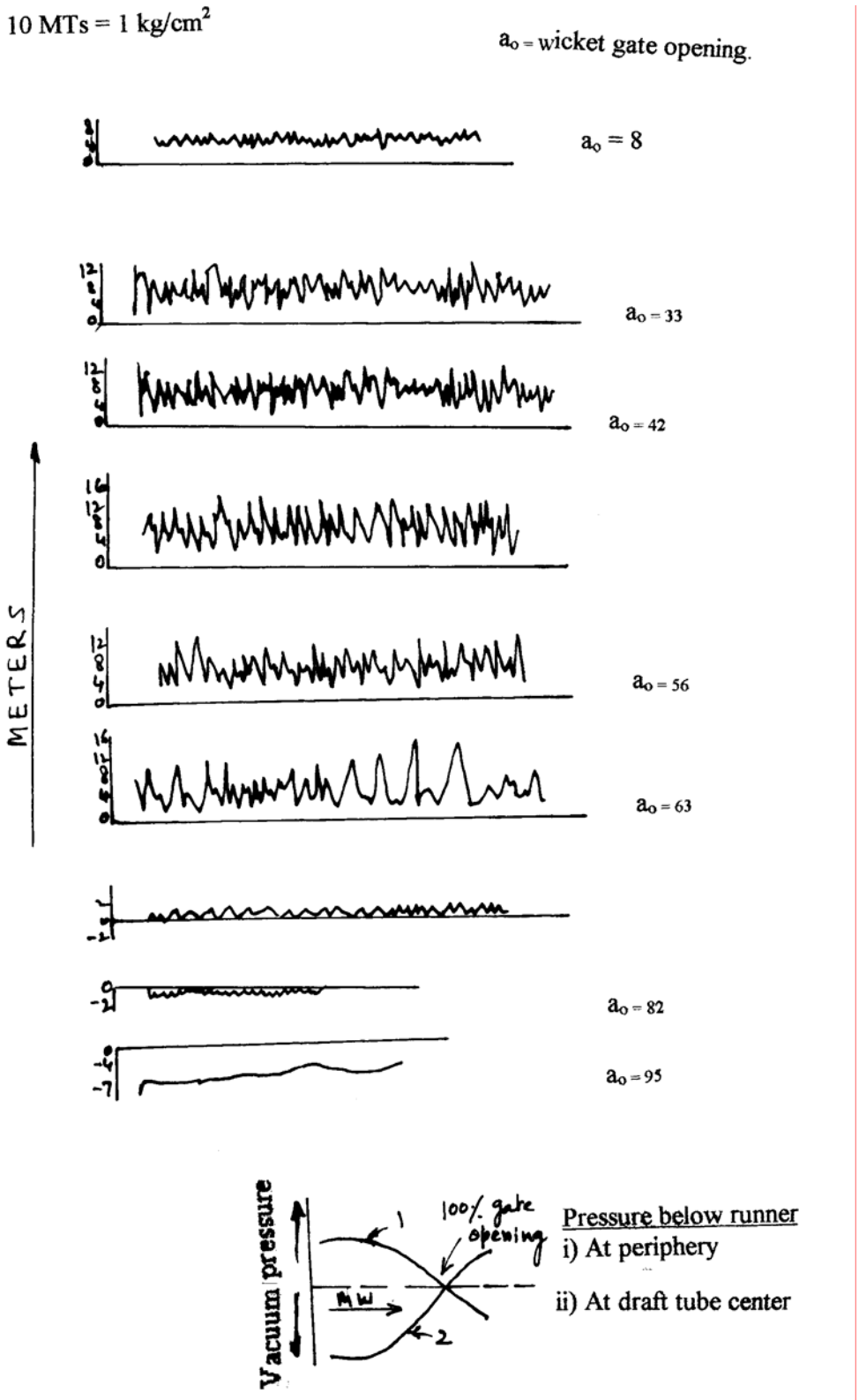


FIG. 11.5 PRESSURE, VIBRATION AT PERIPHERY OF DRAFT TUBE W/O AIR INJECTION

a). Supplementing the aeration by injecting compressed air at 12 Kg/cm^2 through runner labyrinth cooling water pipes meant for synchronous condenser operation of unit. It marginally reduced noise/vibration level.

b) Supplementing of atmosphere air admission through holes drilled in gland support plate on top cover & connecting suitably to pipe work but this was not found successful.

c) The draft tube cross pieces originally located at approximately 1.0 D below the runner for breaking the vortices rope was modified by welding & terminating it first up to 150 mm & then up to 500 mm below the runner. The latter proved successful & was finally adopted.

ii) BHEL also suggested avoiding operation of machine at 30-40 MW (Below 70% rated capacity) at heads below 65 Mtrs (Normal head is 68 mtrs.) & to load the machine quickly through this trouble zone. Implementation of above along with recommendation at para 2.1 reduced pressure pulsation, noise level & eliminated cracks.

4.2.4 BHAKRA LEFT BANK POWER PLANT

i)The pressure pulsation at draft tube & top covers was measured at the time of up-rating from 90 to 108 MW machines of Bhakra left Bank power plant. The pressure pulsations in top cover were very low in magnitude but not readable from record. The pulsation at draft tube at 124.14 M head were as below (fig 11.6).
i)

Load MW	Kg /cm^2	% age pulsation	Total
No load	-0.00 to +0.44	-0 to +3.5	3.5
10	-0.00 to +0.66	-0 to +5.2	5.2
20	-0.38 to +0.88	-3.0 to +7.1	10.1
30	-0.33 to +0.94	-2.6 to +7.6	10.2
40	-0.61 to +1.19	-4.9 to +9.6	14.5
50	-0.19 to +1.33	-1.53 to +10.7	12.23
60	-0.00 to +0.61	-0 to +4.92	4.92
70	-0.00 to +0.38	-0 to +3.06	3.06
80	-0.00 to +0.22	-2.18 to 1.77	3.95
90	-0.19 to +0.00	-1.53 to 0	1.53

Total value is within 12% but more than $\pm$ 6% in the range of 20 to 50 MW. It is very stable in the range of 70 to 90 MW.

ii) The pressure (Kg/cm^2), vibrations (Microns Pk-Pk) & noise level (dbA) were measured on unit no. 2 during 1995 at a head of 410 feet as a residual life assessment studies. The results obtained are as follows—

The below table indicates that draft tube pressure & noise is very high up to about 60 MW (67% of capacity) & stable after 70 MW (78% of capacity). During 1982, instead of absolute pressure, pressure pulsation was measured & the same was found high up to about 60 MW load. Efforts are there fore made by plant engineers to run the machines between 70 & 105 MW (78 to 110% capacity) unless dictated by system requirements.

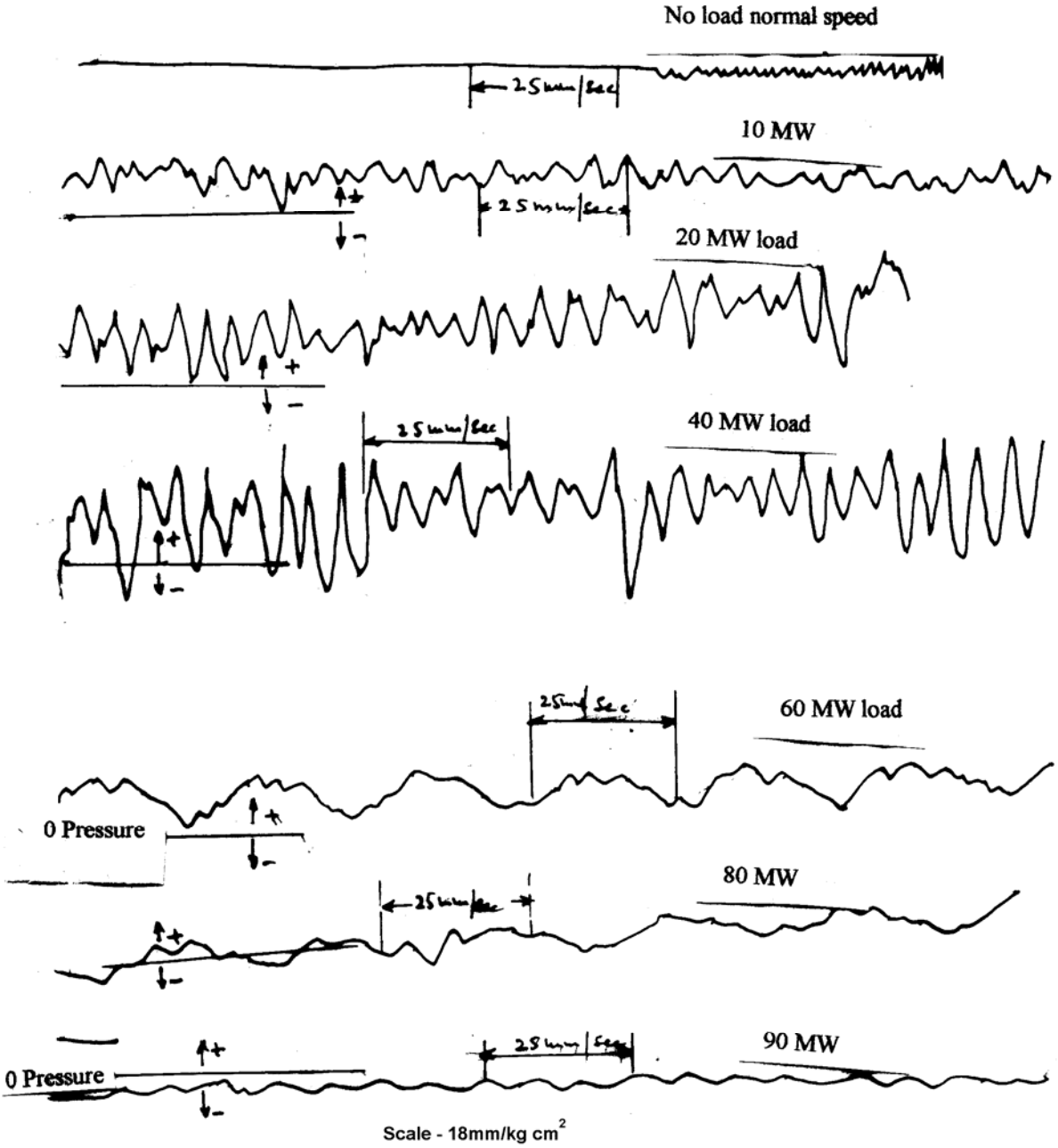


FIG. 11.6. PRESSURE PULSATION MEASUREMENT ON HYDRO MACHINE NO. 4, PP2 BBMB, AT THE TIME OF CARRYING OUT UPRATING STUDIES (DRAFT TUBE)

Load	Pressure		Noise Level			Vibration
MW	Draft tube	Spiral Case	TGB	Spiral	Draft tube	TGB
Stand Still	0.02	12.2	-	-	-	--
No load	0.2	2.15	90	85	92	137
15	0.3	12.05	92	86	97	111
50	0.4	12.00	92-98	90-91	96-103	121
60	0.25	12.00	99-101	90-93	96-103	NA
70	0.02	12.00	85. 5	89.5	103	95
90	0.02	11.85	100	90.5	104	116
100	0.02	11.80	97	87.5	102	110
105	0.02	11.75	96.5	87.0	100	109

4.3 MATCHING OF NATURAL FREQUENCY WITH VON-KARMAN VORTEX FORMATION

When water flow leaves past the trailing edge of an obstacle like runner blade, a disturbance called Von-Kerman vortices shedding is generated. The frequency of shedding of this is dependent upon a no. of factors such as outlet edge geometry, thickness, flow velocity etc. If the frequency of von Kerman vortices shedding comes close to natural frequency of obstacle, the vibration level increases. If it coincides, then it increases very much. This seriously damages the trailing edge of obstacles like runner blade, guide vanes etc. It is reduced by modification of outlet edge geometry & thickness of runner blade or vane etc. The phenomenon is illustrated with the help of following case studies

4.3.1 CASE STUDIES:-

i) PONG POWER HOUSE – BANG TEST

Due to damage to runner blades of Pong Power house (as explained in para 4.1.1.), the frequency response test was carried out on runner blade of unit no. 3 at site in situ condition with the help of BHEL Engineers. The runner was subjected to a bang and the decaying time signal was studied for spectrum analysis. The spectrum indicated the natural frequency of the component under test—Runner in this case. It was noted that the band of natural frequency of the runner blades was different from the von-Kerman vortices shedding frequency & as such this phenomenon was not held responsible for the damage to runner blades.

ii) BHAKRA RIGHT BANK POWER HOUSE

Similar tests to investigate reasons for excess cracking of blades of Right bank power house Bhakra (as described in para 4.1.2.) were also got conducted through different agencies as follows:

i) Department of Mechanical & Industrial Engineering, University of Roorkie. They found 12 resonances in the range of up to 1000 Hz. The resonant frequency was also found different from blade to blade & from runner to runner. They presumed that crack development was due to resonance of runner blades with the shedding frequencies of Kerman Vortices

ii) Department of Mechanical Engineering IIT New Delhi. They tested on model, spare runner lying at site by a computer programme based on Finite Element method & also on a running machine & concluded as follows

a) Natural frequencies of blades of spare runner indicate possible resonance with nozzle pass harmonics & vortices shedding frequencies.

b) The runner blade, which was tested under actual operating conditions showed four dominating frequency contents. This is due to wide band frequencies of fluid excitation.

c) Vibrations are maximum at a part load of 60 MW. These start reducing above head of 400 feet & become minimum at about 470 feet.

d) Modifications already carried out on machine no. 6 & 10 have reduced vibrations.

After carrying out modifications as explained in para 4.1.2. (ii), no problem has been experienced. Since natural frequency of runner blade goes on changing due to its material loss & weld repairs, this test along with others should be repeated whenever cracks starts increasing without any other reasons.

4.4 EXCESS WEAR, TEAR & IN-BALANCE OF RUNNER

Vibration & noise level tend to increase due to excess wear & tear & hydraulic in-balance. The hydraulic in-balance occurs due to rotor in-balance & uneven wearing of gland seal rims. The increased vibrations then adversely effect the worn out portions of blades causing fatigue failure & resulting in cracks & even washing away of pieces. This disturbs the water flow causing still more vibrations, turbulences, eddy currents etc. resulting in much more erosion & vibrations. This is illustrated with the help of following case studies

i) During Oct./Nov. 1993, some abnormal vibrations & noise was observed on unit no. 2 of Ganguwal powerhouse. On opening, a 33" long crack was found on one blade. After repair, both defects disappeared.
ii) Due to silt erosion, labyrinth seal clearance of Dehar power house machines increases. It increases crown pressure, temperature of the thrust bearings & vibrations. Now holes have been drilled to quickly drain out water to reduce crown pressure. This decreased vibrations. Vibrations further get reduced after carrying out weld repairs.
iii) On all silt effected power plants of N.H.P.C., the vibration level increases within 3- 6 months of complete overhaul of machine due to their wear & tear by silt.

5) MECHANICAL REASONS FOR EXCESS VIBRATIONS

Causes of Excess Vibration
i). Misalignment & loss of centricity and imbalance.
ii). Self excited vibration.
iii). Touching of rotating parts with stationary parts.

5.1 MISALIGNMENT AND LOSS OF CENTRICITY

Misalignment due to any reason viz bending of shaft due to deformation, loosening or unequal tightness of coupling bolts, maladjustment of thrust bearing, excess & wrong air gaps of guide bearing etc. causes excess vibrations which results in increased shaft stresses & makes load on bearing segments unequal. Excess stresses may decrease life of components & cause mechanical failure. The source can be identified from frequency of vibrations & operational behavior (Para.7). Some case studies of misalignment are as below

5.1.1 MACHINE NO. 2 BHAKRA LEFT BANK POWER HOUSE

During 11/95, Shaft vibrations of unit no. 2 of left bank power house were got measured from BHEL in connection with life assessment studies of Bhakra left bank machines as follows:
i) They used non-contact probes, which work on eddy current principal & held them at calibrated distance from shaft. Two probe perpendicular to each other were installed at TGB, LGB & UGB. With this, they measured DC gap voltage, overall shaft vibrations (Pk-Pk microns) & phase angle from stationary condition to 105 MW. Results as per table 5.1.1 were obtained-

REMARKS ON ABOVE RESULTS BY BHEL

When the machine is in idle condition, the gap voltage is kept around 9 volts. When the machine is spun, the voltages are as under:

	Channel – I	**Channel – II**
T.G.B.	7.35	8.87
L.G.B.	7.75	8.91
U.G.B.	9.65	12.2

It indicates that the shaft has occupied a different position & continues to be more or less in the same position throughout the load regimes as normally found in healthy machines. Further gap voltage at 100 MW is as follows:-

	Channel – I	**Channel – II**
T.G.B.	7.51	8.34
L.G.B.	7.56	9.17
U.G.B.	9.82	12.7

This shows that U.G.B. is having a much higher clearances & are not uniform also. This indicates a severe rigid body movement of shaft at U.G.B.

The shaft vibrations at 100/105 MW load are as under

T.G.B.	78-110
L.G.B.	196-241
U.G.B.	229-318

BHEL further opined that these values are inclusive of shaft run out. Levels as such are not on very high side. The U.G.B of generator is showing an orbit, which is indicative of severe misalignment & rubs. BHEL suggested that the bearing at U.G.B may be checked. It may also be checked whether any pad or its support is loose, causing impact with other components. The bearing alignment & clearances may be checked. The three bearings may have to be recentered.

The machine was subsequently overhauled & realigned. The vibrations came down (3.2.2(iii)), but were still not within ideal limits.

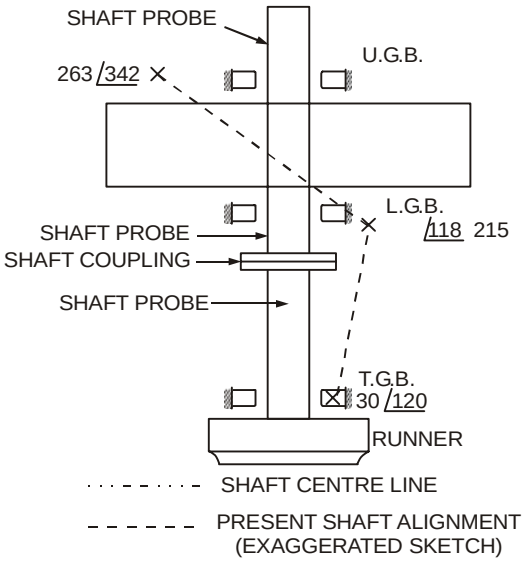


FIG. 11.3 - CASE STUDY OF ALIGNMENT OF MACHINE NO. 2 PPI (N.T.S.)

ii) CASE OF MEASUREMENT OF ACCELERATION

Bracket vibration (overall acceleration) in M/sec^2 of the above machine (machine no. 2 PPI) was also measured & the results are as below:

Load MW	UGB		LGB		TGB	
	Axial	Radial	Axial	Radial	Axial	Radial
No load	-	-	-	-	6.11	12.8
15	1.48	0.463	0.57	5.75	1.35	5.26
50	0.868	1.62	0.818	0.935	3.08	1.61
70	0.801	0.32	0.659	0.852	2.57	1.34
90	1.04	0.364	0.559	1.4	4.55	1.72
100	0.876	0.886	0.856	0.382	3.82	3.34
105	0.759	0.417	0.993	0.947	0.411	0.374

REMARKS ON THE RESULTS BY BHEL

U.G.B. The spectrum indicates electrical problems like non-uniform air gaps, high dynamic eccentricity of generator rotor.

L.G.B. The spectrum indicates electrical problems as described above. It is required that generator stator/rotor air gaps may have to be checked w.r.t commissioning values.

T.G.B. The spectrum indicates non-symmetrical flow passages, which could be due to extensive weld repair of blades.

5.2 SELF-EXCITED VIBRATIONS

Self-excited vibration appears due to following reason
i). Inadequate lubrication due to defective design or low level or improper lubricant.
ii). Touching of shaft with bearing metal, excess or wrong gaps of guide bearings.
iii). Inadequate support of guide bearing supports.
iv). Touching of rotating parts with stationary parts.
v). Deviation of blade angle from optimum position incase of Kaplan turbine.

5.3 UNBALANCING OF ROTOR/RUNNER

Due to large mass & heavy weight, the rotor's flywheel effect is large. If the main mass of the machine i.e. its rotor or runner is not balanced, then it would rotate in its bearing with unbalanced centrifugal force, which would give rise to excessive run out. This excessive run out would result in opening & damage to bearings, sparking at commutator & slip ring, change in air gap & resultant excessive stresses. This imbalance may be due to manufacturing / assembly defect; improper maintenance & operation; bending of shaft; non uniform weights of rotor poles & their improper fixing, unequal wear of runner blades; inaccuracies in manufacture of runner blades, exciters, heavy weld repair etc. etc. During operation, runners in silt effected power plants get highly eroded, requiring heavy weld repairs. This generally results in unbalancing of runner by hundreds of kilograms. In Kaplan turbine un-balancing may occur due to blade angle inaccuracies. Such inaccuracies should not be more than $\pm$ 0.5% between different blades.

Efforts should therefore be made to make rotating mass uniform. The balancing of runner is carried out at manufacturer's shop. It may not be possible to achieve perfect uniformity in a large & heavy sized hydro machine in power plant but as much improvement as possible should be tried. Due to heavy weld repairs, facilities for runner balancing should exist in all silt effected power plants. The size & weight of generator rotor being very high, its balancing cannot be done at manufacturer's shop & is therefore carried before commissioning in power plant.

5.3.1 CASE STUDIES:

i) UNIT NO. 5 OF DEHAR POWER PLANT:-Vibrations (microns Pk-Pk) of machine no. 5 of Dehar power plant at the time of commissioning before & after adding balancing weights are given below

Load	L. M. B *				Top Bearing				Turbine bearing after balancing	
	D/S Side		S/B Side		D/S Side		S/B Side			
	Before Bal. **	After Bal.	Before Bal.	After Bal.	Before Bal.	After Bal.	Before Bal.	After Bal.	D/S	S/B
MECH.	179	120	74	20	21	31	NA	39	39	20
11 KV	245	90	61	15	20	49	NA	10	37	29
165 MW	-	45	-	8	-	42	-	3	36	29

** Balbalancing.
* LMB - Lower Main Bracket
Note: 66 Kg, 25 Kg. & 70 Kg. weights were added at three locations.

ii) UNIT NO. 6 OF DEHAR POWER PLANT

Vibrations (microns Pk-Pk) of machine no.6 of Dehar power plant at the time of commissioning before & after adding balancing weights are given below

Load	L. M. B				Top Bracket		Turbine Bearing	
	D/S Side		S/B Side		After balancing		After Balancing	
	Before balancing	After Balancing	Before balancing	After Balancing	D/s Side	S/B Side	D/S	S/B
MECH	355	68	98	73	-	-	-	-
11 KV	347	14	215	34	-	-	-	-
165MW	-	40	-	10	18	24	18	22

Note: 59.3 Kg, & 49.7 Kg. weights were added at two locations on rotor for balancing.

iii) UNIT NO.2 OF DEHAR POWER PLANT:-Vibrations of machine recorded after renovation on 30.9.93

Load	U.G.B		L.G.B	
Mechanical run	260	180	200	260
11 KV	300	270	180	290
140 MW	280	210	170	210
160 MW	240	180	120	170

NOTE: Due to paucity of time, balancing was not done.

iv) MACHINE NO. 4 OF BHAKRA LEFT POWER HOUSE The displacement of this machine was excessive. Repeated efforts were made to reduce the same. Values recorded are given below:

	1/93 Before alignment	3/93 after alignment	1/94 after capital mtc & proper centering	After balancing
Throw	40 Thou.	7 Thou.	5 Thou.	
Off center	20 Thou.	2 Thou.	1 Thou.	
Horizontal displacement				
Excitor panel	0.4 mm	0.4 mm	0.4 mm	0.2 mm
T.G.B.	-	0.08 mm	0.06 mm.	
Vibration				
D.T. Manhole	-	18 Micron	20 Microns	
Excitor panel		11-26 Microns	20-33 Microns	

NOTE: Displacement of 0.4 mm was causing excess sparking of carbon brushes. All other efforts to improve sparking failed. Best efforts to reduce horizontal displacement also failed. Rotor balancing was got done with the help of BHEL by adding weights at three angles & the run out came to 0.2 mm.

5.3.2 VIBRATION OF SOME OF OTHER MACHINES

i). Vibration of new machine no. 2 Ganguwal Power House 27.5 mw propeller turbine - 1/98

Sr No.	Load MW	Shaft					Housing	
		U.G.B		L.G.B	T.G.B		U.G.B	T.G.B
		D/S	90^0		D/S	90^0		
1	167 rpm	69*L*96	56*L*14	4*L*133	25*L*361	50*L*345	6*L*122	10*L*255
2	Full excit.	63*L*94	60*L*15	92*L*135	15*L*146	80*L*369	12*L*273	10*L*189
3	10 MW	63*L*88	60*L*4	90*L*130	8*L*140	62*L*340	12*L*251	10*L*189
4	22 MW	63*L*88	60*L*4	92*L*125	20*L*140	65*L*345	12*L*251	10*L*189
5	26 MW	66*L*98	0*L*18	91*L*121	24*L*110	64*L*340	13*L*250	11*L*225
6	27.5 MW	65*L*98	70*L*17	90*L*121	21*L*120	80*L*310	12*L*245	13*L*189

ii) Vibration level (microns rms) at T.G.B. unit no.4 Left Bank powerhouse measured during 1982

Load MW	Amplitude microns	Velocity mm/Sec	Acceleration M/sec^2
NO LOAD	2/5	0.3	0.89
10	2/3	0.22	0.80
20	2/3	0.25/0.28	1.20
30	5/7	0.5	2.00
40	7/15	0.5/0.6	1.80
50	10/20	0.5/0.7	2/2.24
60	3/7	0.45	1.5/1.65
70	4/6	0.46	1.4/1.5
80	4/10	0.45	1.7/1.8
90	5/8	0.55	2.7/3

NOTE: The above values are satisfactory.

iii) Vibration of unit no. 3 Pong Power Plant on 23.2.98 - Microns pk-pk 6 x 60 MW

Sr.No	Load MW	Shaft				Housing	
		G.G.B		T.G.B		G.G.B	T.G.B
		U/S side	Unit 4 side	U/S side	Unit 4 side	U/S side	Unit 4 side
1	106 rpm	4*L*373	25*L*271	47*L*329	32*L*232	4*L*265	2*L*236
2	63 rpm	15*L*299	12*L*200	50*L*337	33*L*246	6*L*375	4*L*217
3	Full exct.	16*L*35	17*L*293	52*L*333	46*L*241	17*L*70	7*L*132
4	15 MW	8*L*22	8*L*300	55*L*333	48*L*246	11*L*76	7*L*151
5	30 MW	4*L*386	4*L*308	54*L*334	48*L*248	14*L*68	7*L*171
6	45 MW	2*L*127	2*L*267	55*L*334	48*L*240	5*L*82	10*L*167
7	60 MW	6*L*164	3*L*35	56*L*335	47*L*245	14*L*63	9*L*153
8	66 MW	7*L*158	5*L*33	56*L*336	46*L*243	11*L*65	12*L*143

iii). Vibration of different machines of Bhakra Power Houses after rotor balancing measured from 13.1.98 to 18.1.98 in Microns pk-pk. instrument IRD 885

MC No.	Load MW	SHAFT						Housing		Diametrical Bearing clearance mm		
		U.G.B		L.G.B		T.G.B		U.G.B	L.G.B.			
		D/S	W/S	D/S	W/S	D/S	W/S			UGB	LGB	TGB
1	108	13*L* 266	22*L* 167	99*L* 90	118*L*7	20*L* 87	24*L*10	11*L*90	--	0.6	0.55	0.4
2	106	64*L* 95	55*L* 357	186*L* 76	139*L* 334	32*L* 67	--	24*L* 356	--	0.4	0.55	0.6
3	106	43*L* 333	49*L* 262	179*L* 24	102*L* 290	25*L* 35	32*L* 261	--	7*L* 294	0.6	0.77	0.4
4	90	260*L* 243	168*L* 114	29*L* 311	56*L* 234	--	39*L* 300	17*L* 344	16*L* 352	0.4	0.60	0.7
5	105	75*L* 250	122*L* 177	35*L* 255	40*L* 182	38*L* 248	45*L* 143	22*L* 184	--	0.6	0.43	0.4
6	130	269*L* 142	160*L* 34	103*L* 337	153*L* 257	146*L*2	-	135*L* 328	--	0.4	0.40	0.4
9	140	88*L* 208	150*L* 116	166*L* 235	210*L* 138	65*L* 257	--	--	--	0.4	0.40	0.4

NOTE : Vibration of Machine No. 4, 6, 9 are high as compared to other machines.

iv) Vibration of Machine No.1 of Bara Suil at the time of commissioning (Microns but not in Pk-Pk)

	At No Load					At 1.5 MW Load				
	At upper bracket	**Oil Bath**	**Lower bracket**	**Stator Frame**	**TGB**	**At upper bracket**	**Oil bath**	**Lower bracket**	**Stator Frame**	**TGB**
Radial	6	4	3	6	6	16	11	11	8	12
Vertical	3	10	2	4	5	9	18	5	5	9
Tangential	3	4	4	7	10	11	10	10	9	11

6) DYNAMIC STRESSES DUE TO ELECTRICAL FAULTS:-

6.1 The impact of electricals in generator remains mostly confined to generator parts viz. stator & spider & to a very small extent on runner due to absorption of shocks by large mass of generator. It causes unbalanced magnetic forces due to:-

i). Non uniform stator rotor air gap
ii). Failure of insulation in any field pole.
iii). Asymmetrical loading of generator.
iv). Loose winding, broken or loose bracing ties & high partial discharge.

6.2 The reasons are self explanatory from the names. However following is supplemented--

i). Non uniform air gap causes large radial forces to the rotating element. Utmost care should be taken during assembly of stator & rotor cores that true circularity in both cases is achieved. Similarly during alignment, uniform air gaps should be obtained. Designers have calculated that 10% differences in gaps create force equal to about half the weight of rotor. The effect of non-uniform stator-rotor air gap has also been discussed in chapter 2 para 1.1.4. & one case study discussed in chapter 5 para 2.8.3.1(iii).Remedial measures to remove discrepancies have also been discussed in chapters on stator & rotor maintenance.

ii) Due to failure of insulation in the rotor poles, unequal current flows in these pole windings resulting in creation of non-uniform magnetic field & causing increase in vibrations. Remedial measures have been discussed under para 2.1.2 & 2.2 chapter 7.

iii) Ideally current flowing in all the three phases should be equal. However due to system requirement, these may become unequal & result in variable rotating magnetic flux & hence magnetic unbalance & excess vibrations. Maximum difference in currents in phases should be restricted to 20%.

7) ANALYSIS OF VIBRATIONS:

It can be done from the frequency of the vibration as vibration exciting frequency & machine behavior due to different reasons is different as indicated in following tables-

7.1 ANALYSIS OF MECHANICAL VIBRATIONS

Watch the behavior of machine from start to full speed without excitation & load & observe increase / decrease of vibration as follows –

S. No.	NATURE OF VIBRATION	PROBABLE CAUSE	REMEDIAL MEASURES
1.	Vibration increase is proportional to square of speed	Mechanical unbalance in runner or rotor	Dynamic balancing of unit
2.	Vibration increase is proportional to speed	Looseness of rotating Components	Check tightness of coupling bolts, rotor rim & rotor poles
3.	Run out of shaft & vibration of units independent of speed	Improper alignment, incorrect bearing gaps	Re center & realign the machine

7.2 ANALYSIS OF MAGNETIC UNBALANCE –

Watch the behavior of machine without load but with increase in excitation & record increase/decrease in vibrations & take remedial measures as described below –

S. No.	NATURE OF VIBRATION	PROBABLE CAUSE	REMEDIAL MEASURES
1.	Vibration increases with increase in excitation	i) Inter turn short in pole coils	Attend as per chapter 7 para 2.1.2.& 2.2
		ii) Variation in air gap	Values as per chapter para 3.5. (vii)

7.3 HYDRAULIC VIBRATIONS

If values of vibrations as per above paras are satisfactory, then put load & increase it slowly. Increase in vibration is then due to hydraulic causes as tabulated below:

S. NO.	NATURE OF VIBRATION	PROBABLE CAUSE	REMEDIAL MEASURES
1	Vibration & noise level increases at part load & over load	i). Vortex formation in draft tube ii). Cavitations	i). Avoid running at part load /over load. ii). Consult manufacturer for breaking vortex formation by admitting air or by other means.
2	Vibration increase with increase in load	Design defect in runner, matching of natural freq. of turbine with pressure fluctuation freq. etc.	Consult manufacturer for modification of runner.
3.	Excess vibrations at a particular load	Design defect in runner, formation of Kerman vortex	Consult manufacturer for modification of runner

This article is a copy of chapter no. 11 of book "Diagnostic tests, life assessment preventive maintenance, overhauling, refurbishment and up rating of hydro power plants" and compiled by

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